Remarks/Arguments:

Information Disclosure Statement (IDS)

The IDS that was filed on September 21, 2006 has not yet been considered. Consideration of this IDS is kindly requested.

Claim Rejections Under 35 USC §103

Claims 12-22 stand rejected under 35 USC §103(a) as unpatentable over Smith, "Understanding Parameters Influencing Tire Modeling") (hereinafter Smith article) in view of Ono et al., US Patent Application Publication No. 2004/0133330 (hereinafter Ono). Applicants respectfully request reconsideration of the rejection of these claims because the Smith article is not a proper reference for rejecting Applicants' claims.

In the Amendment filed on August 12, 2010, Applicants stated that the Smith article is not a proper reference for rejecting Applicants' claims because the priority application (German Patent Application No. DE 10 2004 177.0), which was filed on March 23, 2004, predates the May 19, 2006 publication of the Smith article. In the Office Action dated October 27, 2010, the Examiner requested that the Applicant submit (i) an English translation of the certified copy of German Patent Application No. DE 102004177.0, and (ii) a statement that the certified English translation is accurate in order to perfect the foreign priority claim to German Patent Application No. DE 102004177.0. **Applicants submit those requested documents herewith.**

Additionally, Applicants note that PCT Application No. PCT/EP2005/051338, to which the instant application claims priority, was filed on March 23, 2005. The filing date of PCT Application No. PCT/EP2005/051338 also predates the May 19, 2006 publication of the Smith article. Applicants submit herewith an English translation of the certified copy of PCT/EP2005/051338 on September 21, 2006. Applicants also submit herewith a statement that the certified English translation is accurate in order to perfect the foreign priority claim to PCT/EP2005/051338.

Applicants reproduce the arguments that were presented in the Amendment filed on August 12, 2010 hereinafter for the Examiner's convenience.

In accordance with M.P.E.P. Section 2128, "[a] reference is proven to be a "printed publication" upon a satisfactory showing that such document has been disseminated or otherwise made available to the extent that persons interested and ordinarily skilled in the subject matter or art, exercising reasonable diligence, can locate it."

The instant application claims priority to German Patent Application No. DE 10 2004 177.0, which was filed on March 23, 2004. Applicants' representative was unable to identify any evidence that the Smith article was made publicly available prior to March 23, 2004. Although the Smith article is marked Copyright 2003, the Smith article was not recorded in the U.S. Copyright Office. Merely marking a copyright date on a document does not confirm that the document has been disseminated or otherwise made available to the extent that persons interested and ordinarily skilled in the subject matter or art, exercising reasonable diligence, can locate it. Additionally, according to web.archive.org, the earliest Internet posting of the Smith article on the website of Colorado State University was May 19, 2006. Because German Patent Application No. DE 10 2004 177.0, which was filed on March 23, 2004, predates the May 19, 2006 publication of the Smith article, the Smith article is not a proper reference for rejecting Applicants' claims.

The Ono patent was cited for the limited purpose of teaching a steering torque detecting portion, and does not overcome the deficiencies of the Smith article.

Reconsideration of the rejection of claims 12-22 and allowance of those claims is respectfully requested.

Conclusion

In view of the remarks set forth above and the enclosed accuracy statements, Applicants respectfully submit that this application is now in condition for allowance, which action is respectfully requested. If the Examiner believes an interview will advance the prosecution of this application, it is respectfully requested that the Examiner contact the undersigned to arrange the same.

Respectfully submitted,

Glenn M. Massina, Reg. No. 40,081
Brett J. Rosen, Reg. No. 56,047
Attorneys for Applicants

GMM/BJR/

Dated: December 27, 2010

Encl: English Translation of the German Priority Document (including an Accuracy Statement)
English Translation of PCT/EP2005/051338 (including an Accuracy Statement)

P.O. Box 980
Valley Forge, PA 19482
(610) 407-0700

The Director is hereby authorized to charge or credit Deposit Account No. **18-0350** for any additional fees, or any underpayment or credit for overpayment in connection herewith.

Legal Advantage

Legal Advantage LLC
4350 East-West Highway – Suite 420
Bethesda MD, 20814

Phone: 301.637.6180
Fax: 800.804.4220

To,
Brett Rosen
RatnerPrestia
Suite 301, 1235 Westlakes Drive,
Berwyn, PA 19482
Email: bjrosen@ratnerprestia.com
Phone: 610-993-4245
Project No. 9903831
Matter No. PCT/EP2005/051338

'CERTIFICATION OF PCT/EP2005/051338'

This is to certify that the attached document in English language is true, accurate translation and has the same meaning as the statements made in the original document of German language to the best of our knowledge and belief.

Executed this 24th day of December 2010.

Hal Pieroway
Translation Department
Legal Advantage LLC
4350 East-West Highway – Suite 420
Bethesda MD, 20814

Tire Lateral Force Determination in Electrical Steering Systems

Field of the Invention:

The present invention relates to a method for determining the tire lateral force in a motor vehicle with an electromechanical or electrohydraulic steering system.

Background of the Invention:

In addition to highly customary ABS brake systems, an increasing number of current motor vehicles are equipped with driving dynamics control systems in order to enhance the active safety of vehicles. Driving dynamics control systems are employed to check and limit yaw movements of the vehicle about its vertical axis. Sensors detect variables predetermined by the driver such as the steering angle, the accelerator pedal position, and the brake pressure, for example. In addition, the lateral acceleration and the rotational behavior of the individual vehicle wheels are measured. The efficiency of the driving dynamics control systems could be increased even further by determining additional variables that influence the dynamic performance of the motor vehicle. Among these variables is e.g. the coefficient of friction of the vehicle wheels on the roadway or the sideslip angle, which indicates the angular deviation of the speed vector from the vehicle's center line.

Abstract of the Invention:

Based on the above, an object of the invention involves disclosing a method, by which at least one additional variable can be determined, which influences the dynamic performance of a vehicle.

This object is achieved using the method described in claim 1. According to the invention, a method is disclosed for calculating the lateral force in a motor vehicle equipped with an electromechanical or electrohydraulic steering system. The method comprises the following steps:

- recording a steering rod force;

- calculating a total restoring torque from the steering rod force, with the said restoring torque comprising a restoring torque generated by lateral force and other restoring torques;
- quantitative determination of the other restoring torques based on measured values;
- subtracting the other restoring torques from the total restoring torque for determining the restoring torque generated by the lateral force; and
- determining the lateral force from the restoring torque generated by the lateral force.

The lateral force at the wheels is a favorable input variable for many driving dynamics control systems. The lateral force can be used to determine the coefficient of friction or to estimate the sideslip angle, for example.

Modern electromechanically or electrohydraulically assisted steering systems or electromechanical or electrohydraulic steering systems, which are mechanically uncoupled by the driver, contain in principle force or torque sensors from which the steering rod force (toothed rack in rack-and-pinion steering) or steering tie rod forces are measured or calculated. The tire lateral forces can be determined from the aforementioned forces. The method of the invention makes use of this sensor equipment in order to define the tire lateral forces.

In an improvement of the invention, a transmission ratio between the steering rod force and the total restoring torque is included in the determination of the lateral force. Suitably, the transmission ratio can be responsive to the steering angle.

Favorably, a kingpin inclination and/or a caster angle are included in the determination of the lateral force.

The other restoring torques that are important for the invention can comprise restoring torques generated by rolling resistance, brake force, driving power, and/or vertical force.

In different embodiments of the method of the invention, the steering rod force can be detected as a force acting on the left and right steering tie rod or as the total steering rod force.

Advantageously, the total steering rod force is calculated from a steering torque generated by the driver, steering amplification, and a steering ratio. It can be provided that a steering-angle-responsive steering ratio enters into the calculation of the steering rod force.

In an embodiment of the invention, the total steering rod force is determined from the motor current and/or the motor position of one or more electric motors of the electromechanical or electrohydraulic steering system.

Thus, the method of the invention can be extended suitably in such a fashion that a sideslip angle and/or a coefficient of friction are ascertained from the determined lateral force.

Short Description of the Drawings:

The drawings schematically illustrate an electromechanical steering system in which a method according to the invention can be implemented. In the drawings:

Figure 1 is a schematic view of an electromechanical steering system;

Figure 2 shows the caster angle and kingpin inclination on a vehicle wheel;

Figure 3 shows the lateral force lever arm on a vehicle wheel;

Figure 4 shows the brake force lever arm on a vehicle wheel;

Figure 5 shows the disturbing force lever arm on a vehicle wheel;

Figure 6 shows the vertical force lever arm on a vehicle wheel and its relation to the kingpin inclination; and

Figure 7 shows the vertical force lever arm on a vehicle wheel and its relation to the caster angle.

Detailed Description of an Embodiment of the Invention:

Figure 1 illustrates the front axle of a motor vehicle and the steering system. A driver directs the vehicle by turning a steering wheel 1 into a desired driving direction. The steering movement of the steering wheel 1 is transferred mechanically to a pinion 3 by way of a steering column 2. Pinion 3 engages a spur rack 4. Rotation of the steering wheel 1 will thus cause the spur rack 4 to move to and fro. Each end of the spur rack 4 is connected to one left and one right steering tie rod 61, 6r respectively, which transmit the movement of the spur rack 4 to front wheels 71 and 7r, respectively, of the vehicle. The suspension of the vehicle front wheels 71, 7r has been omitted in Figure 1 for the sake of clarity. The steering system described thus far is purely mechanical and necessitates great steering forces from the driver under the weight of the vehicle. For this reason, the steering column 2 is additionally coupled to an electric motor 8 in terms of driving, which assists the steering movements of the driver at the steering wheel 1. Although motor 1 is shown in Figure 1 adjacent to the steering column 2, it drives the steering column 2 in reality and acts on the pinion 3. Motor 8 is controlled by a motor control 9 and is fed with energy from battery 11. In addition, the steering column 2 is equipped with a torque sensor 12a and a transducer 12b, which detect the magnitude of the steering torque M_L generated by the driver and sends it to the motor control 9 and to a lateral force calculation unit 13. Furthermore, the motor control unit 9 sends a signal V_L to the lateral force calculation unit 13. The signal V_L describes the amplification of the steering torque M_L generated by the driver. The lateral force calculation unit 13 outputs an output signal representative of the lateral force F_Y that acts on the front wheels 71, 7r.

The mode of operation of the previously described steering system and the method for calculating the lateral force F_Y are described in the following.

Characteristic values for the front wheel suspension have been explained graphically in Figures 2a to 2c for better comprehension of the invention. For the sake of clarity, the characteristic values are illustrated only by way of example of the right front wheel of a vehicle, which is designated by reference numeral 7. Steering movements cause each wheel to swivel around a fixed axis of rotation, which is referred to as steering axis 16. The steering axis 16 firmly connects to the vehicle body at two points E and G. The position of the steering axis 16 relative to a system of coordinates X, Y, Z firmly connected to the vehicle body is described by the following characteristic values.

Figure 2a shows a side view of the wheel 7. The angle between the steering axis 16 and the normal line of the road 17 in the longitudinal plane of the vehicle is referred to as caster angle τ. The distance between the point 18 where the steering axis 16 intersects the roadway 21 and an ideal tire contact point 19 in the vehicle longitudinal plane is referred to as the caster offset $r\tau,k$.

Figure 2b shows a front view of the wheel 7. The angle between the steering axis 16 and the road normal line 17 in the vehicle transversal plane is referred to as the kingpin inclination σ. The distance between the intersection point 18 of the steering axis 16 through the roadway 21 and the ideal tire contact point 19 in the vehicle transversal plane is referred to as the roll radius $r\sigma$.

Furthermore, Figure 2c shows an inclined front view of the wheel 7 in which both the caster angle τ and the kingpin inclination σ are shown.

In electromechanically or electrohydraulically assisted steering systems, the steering torque M_L generated by the driver is measured in order to calculate and adjust the rate of amplification V_L to be provided by the electric motor. Based on the usually steering-angle responsive transmission ratio i_{Ll} (δ) between the steering wheel torque and the total steering rod force $F_{L,sum}$ as well as the steering amplification V_L, the total steering rod force is calculated as follows:

$$F_{L,sum} = M_L \cdot V_L \cdot i_{L1}(\delta) \qquad (1)\ .$$

The total steering rod force $F_{L,sum}$ results from the addition of the forces F_{Lr} and F_{Ll} that act vertically on the steering rod from the right and the left steering tie rods.

In electromechanical or electrohydraulic steering operations, which are uncoupled mechanically by the driver, either both steering tie rod forces are measured separately ($F_{L,r}$ and $F_{L,l}$) or the total steering tie rod force $F_{L,sum}$ is measured or estimated based on the motor current and/or the motor position of the electric motor(s). These forces are e.g. required for the generation of the haptic steering feeling.

The procedure for calculating the single steering rod forces $F_{L,r}$ and $F_{L,l}$ is identical, except for the parameters and the directions of force transferred, and is, therefore, performed using the example of a wheel 7 without wheel indices. The steering rod force F_L compensates restoring torques, which act on the wheel 7 and are generated by different forces. The sum of the restoring torques is referred to by M_z because the total

restoring torque acts along the z-axis of the system of coordinates illustrated in Figure 2.

A second, likewise steering-angle-responsive transmission ratio i_{L2} (δ) acts between the steering rod force F_L and the total restoring torque M_z along the steering axis 16:

$$M_z = F_L \,.\, i_{L2}(\delta) \qquad (2)\,.$$

A restoring torque generated by a lateral force F_y is also comprised in the total restoring torque. The relation between the lateral force F_y and the restoring torque generated by it will be explained in the following.

Figure 3a again shows a side view of the vehicle wheel 7. A lateral force F_y acts upon the wheel 7 at the tire contact point. As the steering axis 16 is tilted in relation to the vertical line by the caster angle τ, the lateral force F_y is applied relative to the steering axis 16 in an offset manner. The distance between the point of application of the lateral force F_y, which corresponds to the tire contact point, and the steering axis 16 is referred to as the kinematic lateral force lever arm $n_{\tau k}$. The lateral force F_y, which is applied to the lateral force lever arm $n_{\tau k}$, generates a restoring torque $M_{z,y}$ according to:

$$M_{z,y} = F_y \,.\, n_{\sim k} \qquad (3)\,.$$

This consideration applies only to the case without movement of the vehicle and without oblique motion of wheel 7.

Oblique motion causes the point of application of the lateral force F_y to displace by the wheel caster behind the middle of the wheel, with the result that the lateral force lever arm is extended. The lateral force lever arm extends in addition to the kinematic lateral force lever arm $n_{\tau,k}$ along the component of the wheel caster $r_{\tau,T}$ perpendicular to the steering axis so that the following applies to the total lateral force lever $r_{\sigma,t}$:

$$r_{\sigma,t} = n_{\tau,k} + r_{\tau,T}{}^{\circ} \cos\tau \qquad (4)\,.$$

The desired lateral force F_y enters into the restoring torque M_z by way of the lateral force lever arm $r_{\sigma,t}$ and the kinematic kingpin inclination σ. The restoring torque generated by the lateral force F_y is designated by $M_{z,y}$:

$$M_{z,Y} = F_y \cdot \cos\sigma \cdot r_{\sigma,t} \qquad (5)\,.$$

Inserting equation (4) into equation (5) results for the restoring torque $M_{z,\,y}$:

$$M_{z,y} = F_y.\, \cos\sigma.\, (n_{\tau,\,k} + r_{\tau,\,T} \cos\tau)\ (6)\,.$$

In addition to the lateral force F_y, further forces act on the steering axis in a torque-generating fashion. In order to separate these torques from the torque $M_{z,y}$ generated by the lateral force, the individual calculation formulas are indicated in the following.

Among the other forces, which act on the steering axis 16 in a torque-generating fashion, is a brake force F_B, which is transmitted from a roadway 21 to a wheel 7. Figure 4 shows a front view of the vehicle front wheel 7. The brake force F_B that is transmitted from the roadway 21 onto the wheel 7 is applied at a distance $r\sigma$ from the intersection point 18 of the steering axis 16 through the roadway 21. The length of the brake lever arm r_b that is normal to the steering axis 16 amounts to

$$r_b = r\sigma . \cos\sigma \qquad (7),$$

and σ indicates the kingpin inclination. In consideration of the caster angle τ, the torque along the steering axis 16 that is generated by the brake force F_B is achieved by:

$$M_{z,B} = F_B . \cos\tau . r_b \qquad (8).$$

Thus, the restoring torque Mz,B generated by the brake force is obtained by:

$$M_{z,B} = F_B . \cos\tau . r_\sigma . \cos\sigma \qquad (9)$$

This calculation applies only to vehicles with an outboard brake. For vehicles with an inboard brake, a disturbing force lever arm r_a, which will be introduced in the following paragraph must be used instead of the brake force lever arm r_b.

As Figure 5 shows, the rolling resistance force and driving power, in contrast to the brake force, does not act via the brake force lever arm r_b, but acts by way of the aforementioned disturbing force lever arm on the steering axis 16 in a torque-generating fashion. The different working levers develop because only force and not torque is transmitted between wheel and wheel carrier for driving power and rolling resistance force F_R. F_R= F_R in the event of intersection in the middle of the wheel (see Figure 5). Thus, the restoring torque $M_{Z,R}$ results from the rolling resistance force F_R:

$$M_{Z,R} = F_R . \cos\tau . r_a \qquad (10).$$

Herein, r_a represents the disturbing force lever arm being normal to the steering axis 16, and $\cos\tau$ takes into account the distribution of forces on account of the caster angle τ. The rolling resistance force F_R can be obtained from the vertical force F_z and the coefficient of the rolling resistance.

A driving power F_A produces likewise by way of the disturbing force lever arm r_a torque M_A along the steering axis 16 according to

$$M_{Z,A} = F_A \,.\, \cos\tau.\, r_a \qquad (11)\,.$$

Furthermore, a vertical force F_z generates a restoring torque, which is significant especially at lower speeds when only minor lateral forces develop.

Due to the kingpin inclination σ, the vertical force F_z scaled with cos τ acts depending on the steering angle δ along with the vertical force lever arm q as a restoring torque as shown in Figure 6:

$$M_{Z,Z1} = F_z \cdot \cos\tau \cdot \sin\sigma \cdot \sin\delta \cdot q \qquad (12)$$

The vertical force lever arm or steering lever arm q is calculated from the tire radius r_{dyn}, the roll radius $r\sigma$ (Figures 2b and 4) and the kingpin inclination σ as follows:

$$q = (r\sigma + r_{dyn} \cdot \tan\sigma) \cdot \cos\sigma \qquad (13)$$

The restoring torque is calculated with the vertical force lever arm as follows:

$$M_{Z,Z1} = F_z \cdot \cos\tau \cdot \sin\sigma \cdot \sin\delta \cdot (r\sigma + r_{dyn} \cdot \tan\sigma) \cdot \cos\sigma \qquad (14)$$

The geometric ratios described above are illustrated in Figure 6.

In addition to the torque generated by the kingpin inclination, the vertical force F_z produces another restoring torque $M_{Z,Z2}$ due to the caster angle τ:

$$M_{Z,Z2} = F_z \cdot \sin\sigma . \cdot \mathrm{COS}\,\tau\, \sin\delta . \cdot n_\tau \qquad (15)\,,$$

wherein the caster offset n_τ indicates the distance between the point of application of the vertical force F_z and the point of attachment to the vehicle. The geometric ratios for this situation are illustrated in Figure 7.

The desired lateral force F_y is calculated from the total restoring torque M_z determined by way of the steering rod force FL as follows. It applies that the total restoring torque M_z is the sum of the individual restoring torques:

$$M_z = M_{z,y} + M_{Z,B} + M_{Z,R} + M_{Z,A} + M_{Z,Z1} + M_{Z,Z2} \qquad (16)$$

Equation (6) is applicable for the lateral force torque $M_{z,y}$. When inserting equation (6) into equation (16) and rearranging, the following results:

$$F_y = (M_z - M_{z,B} - M_{Z,R} - M_{Z,A} - M_{z,Z1} - M_{z,z2}) \,/\, (\cos\sigma \cdot (n_{\tau,k} + r_{\tau T} \cdot \cos\tau)) \qquad (17)\,.$$

It follows from this equation that the subsequent parameters must be determined in order to achieve the lateral force F_y:

σ: kingpin inclination

τ: caster angle

δ: steering angle

r_σ: roll radius

n_τ: caster offset

r_{dyn} : tire radius

r_a: disturbing force lever arm

$n_{\tau,k}$: kinematic lateral force lever arm

$r_{\tau,T}$: wheel caster

The following variables are measured using the sensors already provided for customary driving dynamics control operations in addition to the aforementioned steering torque M_L, the steering rod force F_L, the steering amplification V_L and the transmission ratios i_{L1}, i_{L2}:

F_B: brake force

F_A: driving power

F_z: vertical force

The totality of parameters and measured quantities eventually permits the determination of the lateral force F_y according to equation (17) as described above.

The invention has been described using the example of an electromechanical steering system; however, it also especially suitable for similar applications in electrohydraulic steering systems.

Patent Claims:

1. Method for calculating the lateral force in a motor vehicle with an electromechanical or electrohydraulic steering system, the said method comprising the following steps:

- recording a steering rod force (F_L);
- calculating a total restoring torque (M_z) from the steering rod force, with the said restoring torque comprising a restoring torque ($M_{z,y}$) generated by lateral force (F_y) and other restoring torques ($M_{z,B}$, $M_{z,R}$, $M_{z,A}$, $M_{z,z1}$, $M_{z,z2}$);
- quantitative determination of the other restoring torques based on measured values;
- subtracting the other restoring torques from the total restoring torque for determining the restoring torque generated by the lateral force; and determining the lateral force (F_y) from the restoring torque ($M_{z,y}$) generated by the lateral force.

2. Method according to claim 1, characterized in that a transmission ratio (i_{Lz}) between the steering rod force (F_L) and the total restoring torque (M_z) is included in the determination of the lateral force.

3. Method according to claim 2, characterized in that the transmission ratio ($i_{L2}(\delta)$ is responsive to the steering angle.

4. Method according to claim 1, characterized in that a kingpin inclination (σ) and/or a caster angle (τ) is included in the determination of the lateral force (F_y).

5. Method according to claim 1, characterized in that the other restoring torques comprise a restoring torque ($M_{z,R}$, $M_{z,B}$, $M_{z,A}$, $M_{z,z1}$, $M_{z,sz2}$) generated by rolling resistance (F_R), brake force (F_B), driving power (F_A), and/or by vertical force.

6. Method according to claim 1, characterized in that the steering rod force is detected as a force that acts on the left and right steering tie rod or as the total steering rod force (F_L).

7. Method according to claim 1, characterized in that the total steering rod force (F_L) is calculated from a steering torque (M_L) generated by the driver, a steering amplification (V_L) , and a steering ratio (i_{L1}).

8. Method according to claim 7, characterized in that a steering-angle-responsive steering ratio (i_{L1} (δ)) enters into the calculation of the steering rod force (F_L).

9. Method according to claim 1, characterized in that the total steering rod force is determined from the motor current and/or the motor position of one or more electric motors (8) of the electromechanical or electrohydraulic steering system.

10. Method according to claim 1, characterized in that a sideslip angle is determined from the determined lateral force (F_Y).

11. Method according to claim 1, characterized in that a coefficient of friction is determined from the determined lateral force (F_Y).

Abstract:

A method for calculation of the lateral force in a motor vehicle with an electromechanical or electrohydraulic steering system is disclosed. The method comprises the following steps: firstly a steering rod force is detected from a total restoring torque is calculated. The total restoring torque comprises restoring torques generated by different forces acting on the wheels. Said restoring torques include a restoring torque generated by lateral force and other restoring torques. The other restoring torques are quantitatively determined on the basis of measured values and subtracted from the total restoring torque, in order to determine the restoring torque generated by lateral force. Finally, the lateral force is determined from the restoring torque generated by the lateral force.
(Figure 1)

Legal Advantage

Legal Advantage LLC
4350 East-West Highway – Suite 420
Bethesda MD, 20814

Phone: 301.637.6180
Fax: 800.804.4220

To,
Brett Rosen
RatnerPrestia
Suite 301, 1235 Westlakes Drive,
Berwyn, PA 19482
Email: bjrosen@ratnerprestia.com
Phone: 610-993-4245
Project No. 9903831
Matter No. DE102004014177

'CERTIFICATION OF THE CERTIFIED COPY OF DE102004014177'

This is to certify that the attached document in English language is true, accurate translation and has the same meaning as the statements made in the original document of German language to the best of our knowledge and belief.

Executed this 20th day of December 2010.

Hal Pieroway
Translation Department
Legal Advantage LLC
4350 East-West Highway – Suite 420
Bethesda MD, 20814

Continental Teves AG & Co. oHG

March 19 2004

GP/GF
P 10910

Determining the tire side force in electrical steering systems

For many driving dynamics control systems, the tire side force of the tires would be a helpful input variable. The side force could be used, for example, to determine the friction coefficient or estimate the side slip angle.

Modern electromechanically or electrohydraulically supported steering systems or steering systems mechanically decoupled electromechanically or electrohydraulically by the driver contain in principle force or torque sensors with which the steering rod force (toothed rack for rack-and-pinion steering) or tie rod forces can be measured or calculated. As a result the tire side force can be determined.

The process described below uses sensors to determine the tire side forces.

In electromechanically or electrohydraulically supported steering systems, the torque applied by the driver T_L is measured in order to calculate and adjust the amplification A_L to be applied by the motor.

With the translation i_{L1} (generally depending on the steering angle) between the steering wheel torque and the total steering rod force $F_{L,sum}$ as well as the steering amplification A_L, the total steering rod force $F_{L,sum}$ is calculated as follows:

$$F_{L,sum} = T_L \cdot A_L \cdot i_{L1}(\delta)$$

The total steering rod force results from the addition of the right and left tie rods applying vertical forces F_{Lr} and F_{Ll} onto the steering rod.

In steering systems mechanically decoupled electromechanically or electrohydraulically by the driver, either both steering rod forces separate (F_{Lr} and F_{Ll}) or, likewise, the total steering rod force $F_{L,sum}$ is measured or estimated from the motor current and/or motor position of the electromotor(s). These forces are necessary, for example, for generating the haptic steering feel.

The approach for calculating the individual steering rod forces F_{Lr} and F_{Ll} is identical with the exception of the parameters and directions of force. It is conducted below using the example of a wheel or wheel indices.

Between the steering rod force F_L and the restoring torque T_Z along the steering axle (vertical axis along which the wheel turns), a second, likewise steering angle dependent, translation takes effect $i_{L2}(\delta)$:

$$T_Z = F_L \cdot i_{L2}(\delta)$$

In this restoring torque, the wanted side force F_Y shrinks over the so-called side force lever arm $n_{\sigma,t}$ and the kinematic king pin angle σ. $T_{Z,Y}$ equals here the **restoring torque through the side force:**

$$T_{Z,Y} = F_y \cdot \cos\sigma \cdot n_{\sigma,t}$$

The side force lever arm $n_{\sigma,t}$ is comprised of a kinematic portion $n_{t,k}$ and a portion, which results from the shift $r_{t,T}$ of the contact point of the side force behind the wheel center when slip angles are present. The latter must then still be converted over the castor angle t on the vertical to the steering axle (see image 1):

$$n_{\sigma,t} = n_{t,k} + r_{t,T} \cdot \cos t$$

The restoring torque through side force is calculated as a result:

$$T_{Z,Y} = F_y \cdot \cos\sigma \cdot (n_{t,k} + r_{t,T} \cdot \cos t)$$

In addition to the side force, additional forces accumulate torque on the steering axle. In order to be able to separate this torque from the torque produced by the side force, the individual calculation formulas to follow are applied.

Restoring torque through braking force:
The braking force F_B takes effect over the so-called braking force lever arm r_b (see image 2), adjusted along the distribution of forces through the castor angle t as torque $T_{Z,B}$ on the steering axle.

$$T_{Z,B} = F_B \cdot \cos t \cdot r_b$$

The braking force lever arm is calculated over the king pin angle σ from the castor radius r_σ and the king pin angle σ.

$$r_b = r_\sigma \cdot \cos\sigma$$

The restoring torque through the braking force is calculated as a result:

$$T_{Z,B} = F_B \cdot \cos t \cdot r_\sigma \cdot \cos\sigma$$

This calculation only applies to vehicles with external brakes. For vehicles with internal brakes, instead of the braking force lever arm, the disturbing force lever arm in the next section should be utilized.

Restoring torque through rolling resistance force:
In contrast to the braking force, the rolling resistance and driving forces do not take effect over the braking force lever arm, but rather accumulate torque on the steering axle over the so-called disturbing force lever arm. The different active levers materialize as a result because, in the case of driving and rolling resistance forces, only a force and not torque is transferred between the wheel and the wheel carrier. A clean cut in the wheel center is $F_R' = F_R$ (see image 3). As a result the restoring torque is calculated through the rolling resistance force:

$$T_{Z,B} = F_R \cdot \cos t \cdot r_a$$

Here r_a is the lever arm vertically standing on the steering axle (see image 3). cos t accounts for the distribution of forces resulting from the castor angle.

Restoring torque through maximum force:
Effecting restoring torque through maximum force is especially relevant at lower speeds (lower side forces).

Due to the king pin angle σ, the maximum force F_Z, which is scaled with cos t, works independent of the steering angle δ with the maximum force lever arm q as restoring torque (see image 4):

$$T_{Z,Z1} = F_Z \cdot \cos t \cdot \sin \sigma \cdot \sin \delta \cdot q$$

The maximum force lever arm or the steering lever arm q are formulated as follows:

$$q = (r_\sigma + r_{dyn} \cdot \tan \sigma) \cdot \cos \sigma$$

The restoring torque calculates with the maximum force lever arm as:

$$T_{Z,Z1} = F_Z \cdot \cos t \cdot \sin \sigma \cdot \sin \delta \cdot (r_\sigma + r_{dyn} \cdot \tan \sigma) \cdot \cos \sigma$$

In addition to the torque caused by the king pin angle, the maximum force acts on the restoring torque due to the castor t (see image 5):

$$T_{Z,Z2} = F_Z \cdot \sin \sigma \cdot \cos t \cdot \sin \delta \cdot n_t$$

Calculation of the side force for a wheel:
Finally, the wanted side force F_Y is calculated from the total restoring torque T_Z determined from the steering rod force:

$$F_Y = (M_Z - M_{Z,B} - M_{Z,R} - M_{Z,A} - M_{Z,Z1} - M_{Z,Z2}) \text{ I } (\cos \sigma \cdot (n_{t,k} + r_{t,T} \cdot \cos t)$$

Claims:
1) Procedure that measures or calculates in an electrical steering system the individual forces or the total force from the right and left wheels that act on the steering axle, whereby these steering rod forces determine the tire side forces.

2) Procedure according to 1), whereby the sum of the steering rod forces measured from the steering torque applied by the driver, the steering amplification, and a steering translation are calculated.

3) Procedure according to 1), whereby the steering rod force from the motor current and/or motor position of the electric motor or electric motors is estimated in steering systems mechanically decoupled electromechanically or electrohydraulically by the driver.

4) Procedure according to 1), 2) or 3), whereby the calculated side forces are utilized for estimating the side slip angle.

5) Procedure according to 1), 2) or 3), whereby the calculated side forces are utilized for estimating the friction coefficient.
Image 1

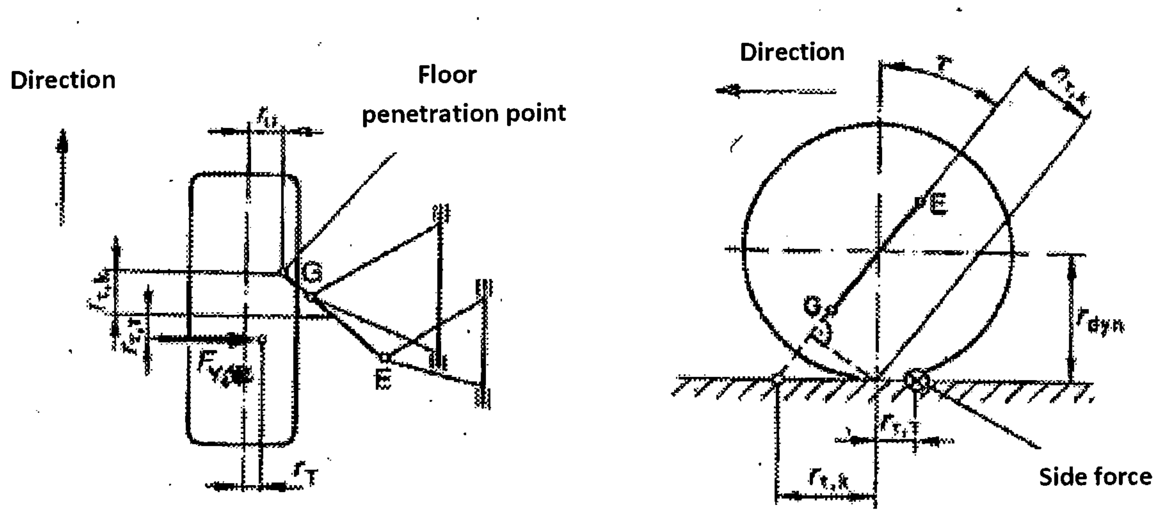


Image 2

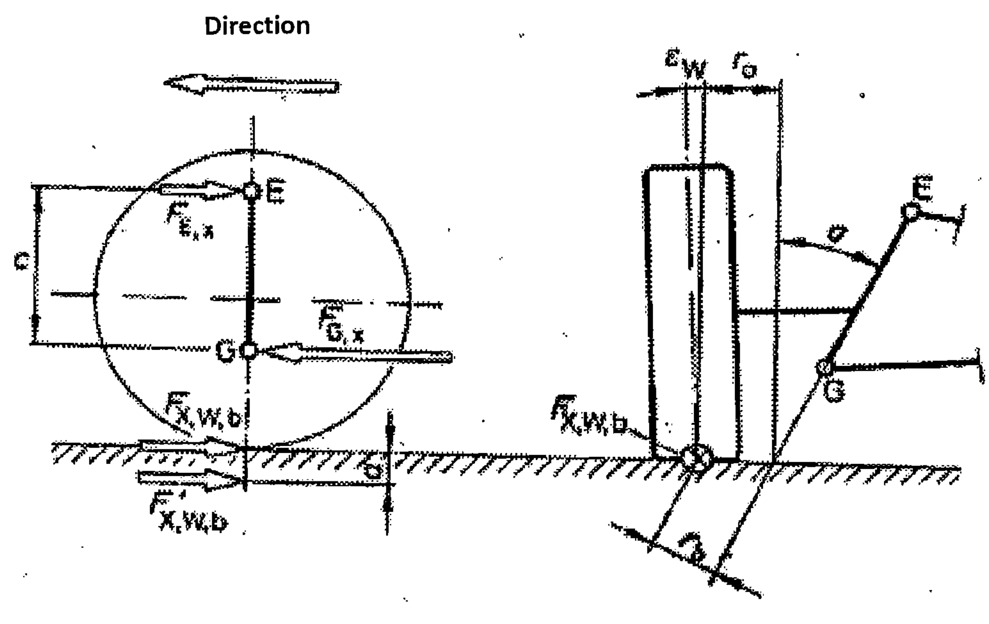


Image 3

Bild 3

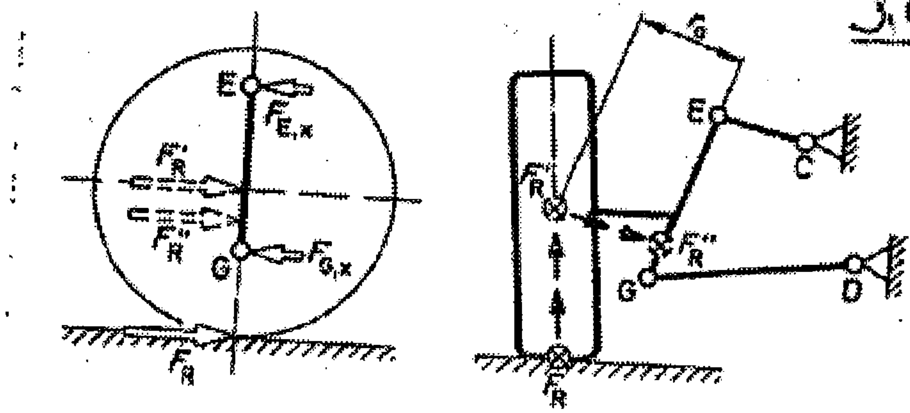


Image 4

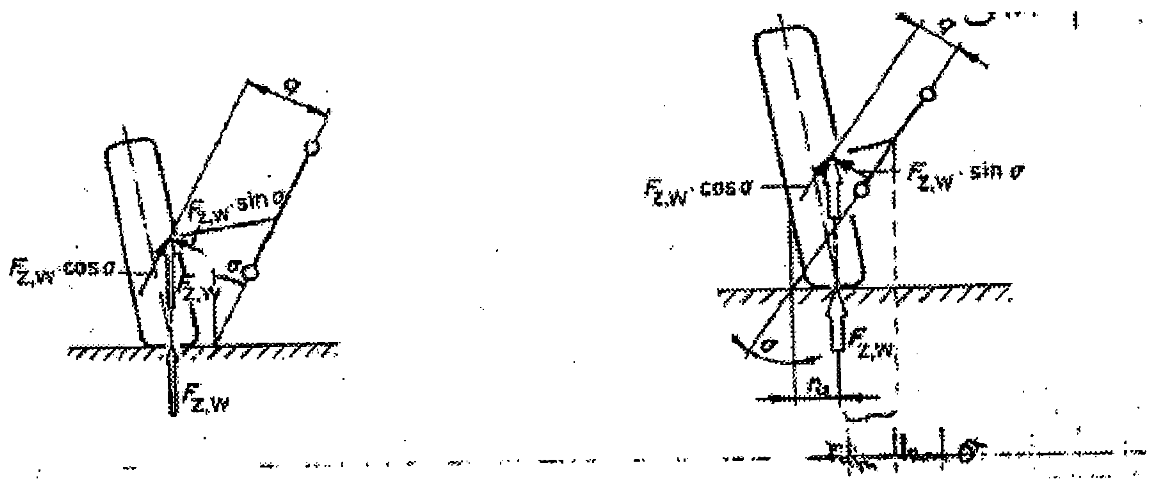
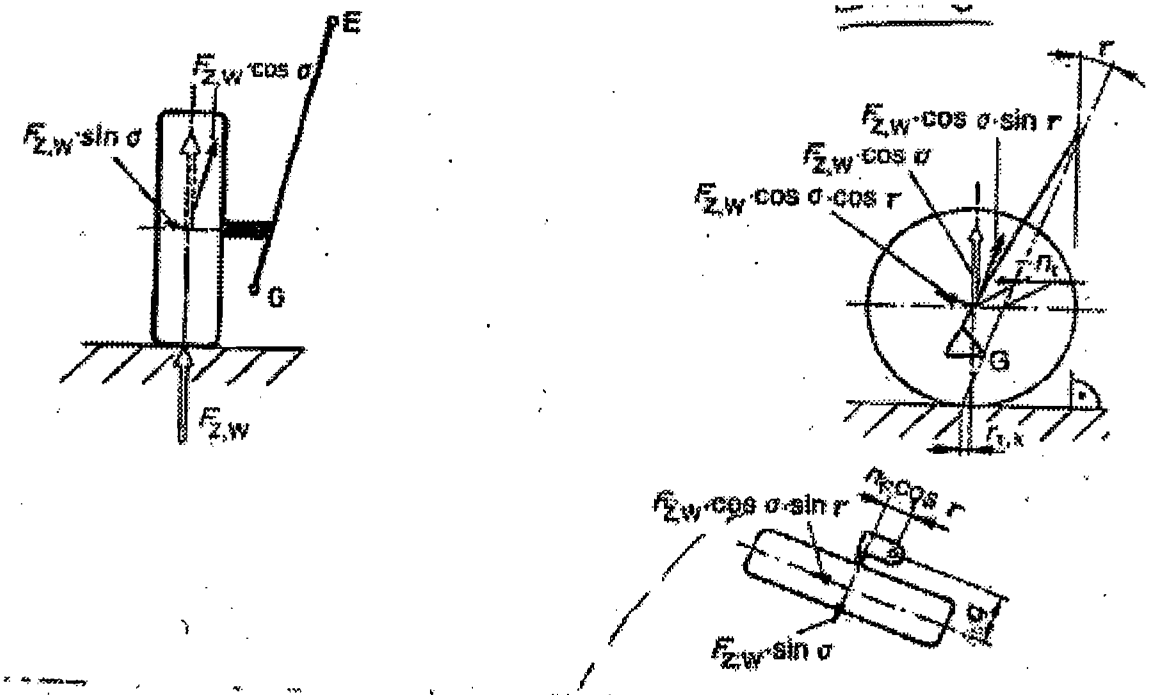


Image 5



E
$F_{Z,W} \cdot \cos\sigma$
$F_{Z,W} \cdot \sin\sigma$
G
$F_{Z,W}$
$F_{Z,W} \cdot \cos\sigma \cdot \sin\tau$
$F_{Z,W} \cdot \cos\sigma$
$F_{Z,W} \cdot \cos\sigma \cdot \cos\tau$
$r_{\tau,k}$
$n_{\tau} \cdot \cos\tau$
$F_{Z,W} \cdot \sin\sigma$

Image 6

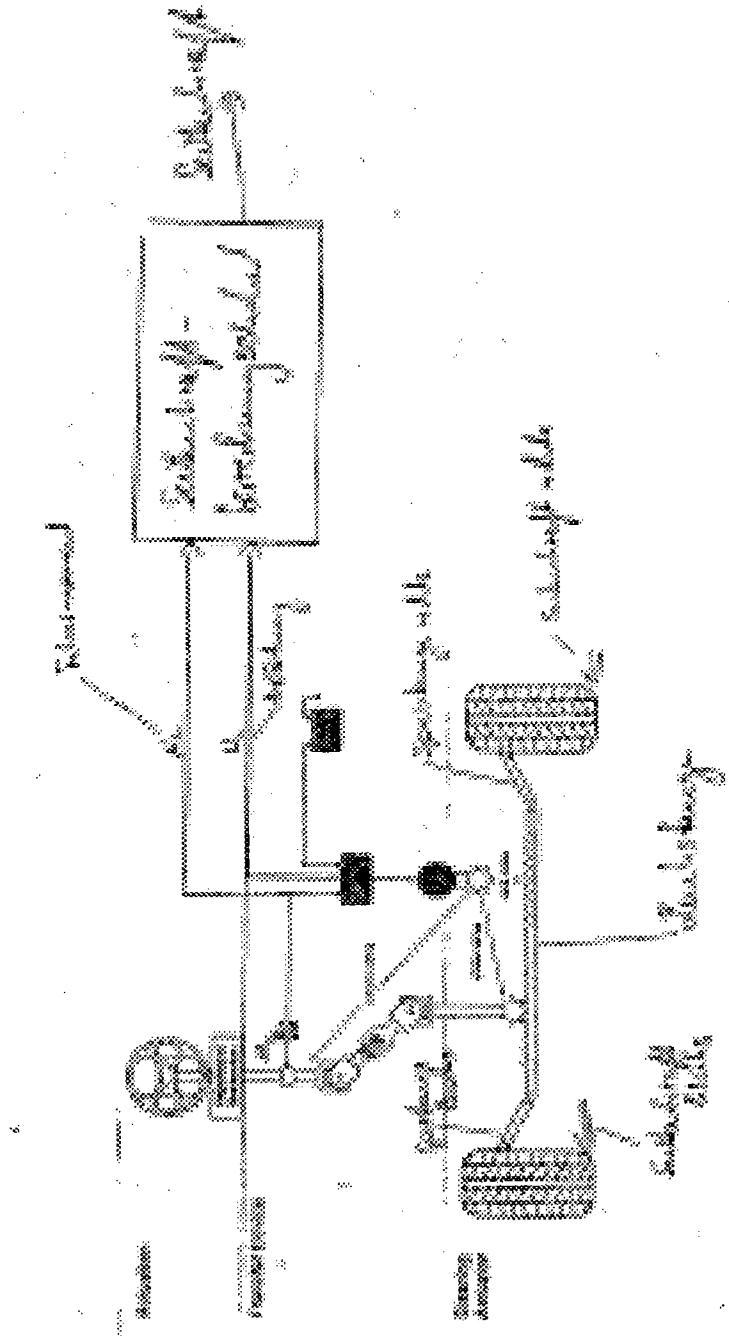


Image 7